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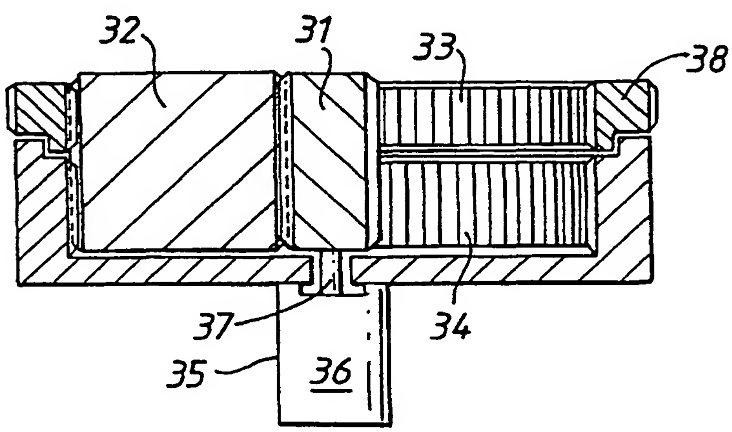

(57) Abstract

There is described an epicyclic speed-reducing gearbox comprising an input shaft (37) driving a sun gear (31), a planet gear (32) engageable with the sun gear (31), a fixed first internally-toothed ring gear (34) having a first number of teeth and capable of meshing with the planet gear, and a second internally-toothed ring gear (33) of the same effective diameter as the first ring gear (34) and capable of meshing with the planet gear (32), but having a second number of teeth different from the first number, the arrangement being such that when a drive input is applied to the sun gear (31), the planet gear (32) travels around the sun gear (31) and the second ring gear (33) rotates relative to the first ring gear (34). If the number of teeth of the first ring gear (34) is less than the number of teeth of the second ring gear (33), then the second ring gear (33) rotates in the same sense as the sun gear (31). If the number of teeth of the first ring gear (34) is greater than the number of teeth of the second ring gear (33), then the second ring gear (33) rotates in the opposite sense to the sun gear (31).

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Reduction Gearbox

The present invention relates to drive transmissions, and particularly concerns a reduction gearbox capable of providing a lower-speed drive output from a high-speed input shaft.

In mechanical devices where a motor is required to drive machinery, it is often the case that in order to run efficiently, the speed of rotation of the motor must be many times the rotational speed of the equipment being driven. Particularly in the case of smaller motors or engines developing lower torque, efficient conversion of electricity or fuel into power may be achieved with motor speeds of from 2 to 3 thousand rpm. In order to convert the lower-torque high-speed output of such small motors into lower-speed high-torque outputs suitable for driving machinery, many types of reduction gearbox have been developed.

In the most commonly used reduction to transmissions, a series of compound gears are meshed together in a gear train. Each compound gear comprises a large radius part and a small radius part, and in the gear train, a driving gear engages the large radius part of the first compound gear, while the small radius part of the first compound gear meshes with the large radius part of the second compound gear. The small radius part of the second

compound gear meshes with the large radius part of a further compound gear, and so on until the final drive is taken from the small radius part of the final compound gear in the train.

The compound gear train reduction transmission has several disadvantages. Firstly, each compound gear has to be supported in a separate bearing, which must resist lateral forces. Secondly, the gear train occupies a considerable space if a large reduction ratio is to be achieved. Thirdly, the large number of meshing gear pairs generates considerable friction and noise.

Planetary or epicyclic gearboxes are known, wherein an internally-toothed ring gear or annulus surrounds a co-axial sun gear, with a number of planet gears held in a rotating planet carrier and meshing both with the sun gear and with the ring gear. A speed reduction may be achieved with such an epicyclic gearbox by holding the ring gear stationary, driving the sun gear, and connecting the load to be driven to the planet carrier. While such a gearbox has coaxial input and output shaft and may therefore be made compact, the need for transmission of torque by the planet carrier can lead to manufacturing difficulties in its production. Furthermore the reduction ratio obtainable with such a gearbox is often insufficient.

An objective of the present invention is to provide a speed-reducing gearbox which is compact and simple in construction, yet provides a high speed reduction ratio.

It is also an objective of the present invention to provide a speed-reducing gearbox having a minimum number of moving parts. Yet another objective of the invention is to provide a gearbox having low tooth loading and thus higher resistance to tooth wear.

According to a first aspect of the invention, a speed-reducing gearbox comprises a pair of coaxially arranged ring gears having the same effective diameter but having different numbers of teeth, the ring gears both being simultaneously engageable by a planet gear.

In an advantageous embodiment, and epicyclic speed-reducing gearbox comprises an input shaft driving a sun gear, a planet gear engageable with the sun gear, a fixed first internally-toothed ring gear having a first number of teeth and capable of meshing with the planet gear, and a second internally-toothed ring gear of the same effective diameter as the first ring gear and capable of meshing with the planet gear, but having a second number of teeth different from the first number, the arrangement being such that when a drive input is applied to the sun gear, the planet gear travels around the sun gear and the second ring gear rotates relative to the first ring gear.

If the number of teeth of the first ring gear is less than the number of teeth of the second ring gear, then the second ring gear rotates in the same sense as the sun gear. If the number of teeth of the first ring gear is greater than the number of teeth of the second ring gear, then the second ring gear rotates in the opposite sense to the sun gear.

It is foreseen that in simple embodiments of the gearbox, a single planet gear will be provided. Advantageously, however, symmetrical arrangements of two or more planet gears may be provided to reduce individual tooth loading and to balance radial forces acting on the sun gear.

The final drive may be taken from the second ring gear by forming it as a pulley, a sprocket, or as a gear with external gear teeth, so that the second ring gear may engage a belt or a chain, or may mesh with a further gear. Alternatively, the second ring gear may be directly attached to a final drive shaft co-axial with the sun gear.

Preferably, the axial length of the sun and planet gears is sufficient to prevent skewing of the axes of the planet gears. In an advantageous embodiment, the combined axial lengths of the two ring gears substantially equal the axial length of the sun and the planet gears.

In the most preferred embodiment, the axial ends of the sun and planet gears are formed with rolling surfaces whose diameter corresponds to the pitch circle diameter of the gear teeth formed on those gears. Likewise, adjacent the gear teeth of the annulus gears there are formed rolling surfaces whose diameters correspond to the pitch circle diameters of the respective annulus gears. The gearbox is arranged such that each planet gear meshes with the sun gear and the rolling surfaces at the ends of the planet gear contact the rolling surfaces at the respective axial ends of the sun gear, while the gear teeth of each planet gear mesh with the gear teeth of both of the annulus gears, and the rolling surfaces at the axial ends of the planet gears engage respectively with the rolling surfaces of the annulus gears. With this arrangement, any radial forces can be transmitted from the sun and planet gears to the annulus gears by means of the rolling contact surfaces, thus freeing the meshing gear teeth from any radial loading.

A two-stage epicyclic reduction gearbox is also contemplated, wherein a sun gear meshes with a number of inner planet gears which in turn mesh with first and second ring gears of equal diameter but differing numbers of teeth. The second ring gear is externally toothed, and serves as a sun gear for a number of outer planet gears which mesh with third and fourth ring gears, co-axial with the first and second ring gears, the third and

fourth ring gears being of equal diameter and having differing numbers of teeth. The first ring gear and one of the third and fourth ring gears is held stationary, and the final drive is taken from the other of the third and fourth ring gears. The number of inner planet gears may be the same as, more than, or less than the number of outer planet gears.

Embodiments on the invention will now be described in detail with reference to the accompanying drawings, in which:

Figure 1 is a perspective view of a transmission incorporating a reduction gearbox according to the present invention;

Figure 2 is an exploded view of the gearbox elements;

Figure 3 is a schematic end view of the gearbox components;

Figure 4 is a transverse sectional view of a second gearbox according to the invention;

Figure 5 is a sectional view similar to figure 4 of a further gearbox, illustrating an alternative output arrangement;

Figure 6 and figure 7 are end views similar to figure 3, showing alternative arrangements of the gears;

Figure 8 is a schematic transverse sectional view of a further arrangement of the gearbox, having two different drive outputs;

Figure 9 is a schematic sectional view of a two-speed transmission incorporating the reduction gearbox of the present invention;

Figures 10 and 11 show alternative drive arrangements for the reduction gearbox;

Figure 12 is a schematic perspective view of a two-stage reduction gearbox;

Figure 13 is a diametral sectional view of the gearbox of figure 12;

Figure 14 is an exploded transverse view showing the components of a further gearbox according to the invention;

Figure 15 is an end view of the gearbox of figure 14; and

Figure 16 is a cross-section taken in the plane XVI-XVI of figure 15.

Referring to figures 1 and 2, there is shown a drive transmission where driving force is transmitted from a motor 1 via a reduction gearbox 2 and a drive belt 3 to a final drive pulley 4, were the drive is transmitted the a final drive shaft 5 to a load 6.

The reduction gearbox 2 comprises a housing 21 to which a fixed annulus gear 22 is mounted. Co-axial with the fixed annulus gear 22 is a moving annulus gear 23 of equal pitch circle diameter to the annulus gear 22. In the embodiment shown, the annulus gear 23 is formed with more teeth 24 on its internal surface than is the annulus gear 22, although the effective diameters of the internal toothed surfaces are the same.

The annulus gear 22 is formed with an external bearing surface 25, and the annulus gear 23 is supported on the bearing surface 25 by means of an internal bearing surface 26 extending along part of the axial length of the annulus gear 23. The external surface 27 of the annulus gear 23 engages the drive belt 3.

A sun gear 28 extends axially from the housing 21 to the distal end of the annulus gear 23. The sun gear 28 is mounted to a motor shaft driven by the motor 1 and supported in a bearing in the housing 21.

Between the sun gear 28 and the annulus gears 22 and 23,

a planet gear 29 is held. The planet gear 29 meshes simultaneously with both of the annulus gears, and in the embodiment shown the gears are all formed with straight teeth.

In the arrangement shown in figure 1, drive belt 3 is wrapped round the outer surface 27 of the annulus gear 23, and transmits the rotation of the annulus gear 23 to the drive pulley 4, which rotates the final drive shaft 5 to supply power to the load 6. In operation, the motor 1 rotates the sun gear 28, which transmits drive to the planet gear 29. Since the planet gear 29 meshes with both the sun gear 28 and the fixed annulus gear 22, the planet gear 29 performs a circulating motion within the fixed annulus gear 22. However, since the planet gear 29 is also meshed with the annulus gear 23, rotation of the annulus gear 23 results because of the difference in numbers of teeth between the annulus gears 22 and 23.

Referring to figure 6, the action of the gearbox is most easily understood. In figure 6 there is shown a planetary reduction gearbox in which a sun gear 31 having ten teeth drives a diametrically-opposed pair of planet gears 32. The planet gears 32 mesh with two co-axial annulus gears, one of which has sixty teeth and the other of which has sixty-two teeth. Since the planet gears have straight-cut teeth, then at the diametrically opposed points T1 and T2 the teeth of the two annulus

gears are held in alignment by the teeth of the planet gears 32. At points T3 and T4, spaced by 90 degrees from the points of alignment T1 and T2, the teeth of the two annulus gears are out of phase with each other, that is to say that the tooth of one of the annulus gears is aligned with a gap between two teeth of the other annulus gear at these points.

If the sun gear 31 is rotated clockwise as seen in the figure, the planet gears will rotate anticlockwise and that engagement with the fixed annulus will cause them to progress round the annulus in a clockwise direction. Since the "points of contact" between the planet gears 32 and the annulus gears must define those locations at which the teeth of the annulus gears are in alignment (as at T1 and T2), and as the planet gears 32 progress round the fixed annulus, the point at which the teeth are in phase also progresses around the circumference of the annular gears.

If the annulus gear with the lesser number of teeth (sixty teeth) is held stationary, then for each revolution of the planet gear 32 round the sun gear 31, the planet gear will engage sixty teeth of the annulus gear with the greater number of teeth. The net effect of one revolution of the planet gear about the sun gear will therefore be to displace the two annulus gears by an amount corresponding to the difference in their

respective numbers of teeth. In this case, the movable annulus gear will rotate relative to the fixed annulus gear in a direction opposite to the direction of progression of the planet gear 32.

If, however, the annulus gear with the greater number of teeth is held stationary, then for each progression of the planet gear 32 round the sun gear 31 the planet gear will engage sixty-three teeth of the movable annulus gear, i.e. the movable annulus gear will be displaced relative to the fixed annulus gear in the same direction as the direction of progression of the planet gear 32.

It has been found in practice that provided the difference in numbers of teeth between the two annulus gears is less than approximately 10%, the planet gear 32 can effectively mesh with both annulus gears simultaneously, using the same tooth profile along the entire axial length of the planet gear. In the example of figure 6, if the annulus gear having 60 teeth is held fixed, six revolutions of the sun gear are necessary to advance the planet gears once round the fixed annulus. During that period, the movable annulus rotates relative to the fixed annulus by an angle corresponding to two of its teeth, a some 11.6 degrees. To effect one complete revolution of the movable annulus, the planet gears will have to make 31 progressions round the fixed annulus, and this movement will require 186 revolutions of the sun

gear. The overall reduction ratio of the epicyclic gearbox shown in figure 6 is a thus 186:1.

The difference in the numbers of teeth between the two annulus gears determines the possible arrangements of planet gears which can be adopted. If the difference in the numbers of teeth is only one, and there will only be one position around the circumference of the annulus rings where the teeth are in alignment between the two rings. It is only possible to mesh a planet gear simultaneously with the two rings at one position, and therefore the gearbox may only comprise one planet gear. This arrangement is illustrated in figure 3, and exemplified in the embodiment of figures 1 and 2.

If, however, the difference in the number of teeth between the annulus gears is two, then as seen in figure 6 there will be two positions where a planet gears can simultaneously engage both annulus rings.

When the difference in the numbers of teeth is three, there are three equispaced positions where the teeth of the annular gears coincide, and thus and arrangement of planet gears such as is seen in figure 7 can be adopted.

As a general rule, the number of planet gears which may be accommodated in the gearbox must be equal to, or a factor of, the difference between the numbers of teeth on

the two annulus gears. In every case, the gearbox will operate with only a single planet gear. It is however preferable to use a symmetrical arrangement of planet gears in order to balance the lateral forces on the sun gear caused by the engagement of its teeth with those of the planet gears.

Figures 4 and 5 are diametral sectional views showing differing arrangements for providing final drive from the reduction gearbox. In the arrangement of figure 4, sun gear 31 drives a single planet gear 32 which engages upper and lower annulus rings 33 and 34, respectively. The lower annulus ring is fixed to the casing 35 of a motor 36 which drives the sun gear 31 via a shaft 37. The upper annulus ring 33 is provided with an external ring gear 38, which may engage further gears to transmit power from the motor 36 to a load.

In the arrangement of figure 5, motor 36 drives a sun gear 31 via a shaft 37, and a planet gear 32 engages upper and lower annulus rings 33 and 34. In this arrangement, however, the upper annulus ring 33 has an upstanding annular flange 39 closed by an end plate 40, from which a stub shaft 41 extends coaxially with the drive shaft 37. The stub shaft 41 may comprise a coupling such as the flange or a dog clutch to transmit rotary movement to a shaft co-axial with the gearbox.

Figure 8 shows a diametral section of a reduction gearbox providing drive outputs at two different speeds. In the arrangement of figure 8, a motor 50 drives a sun gear 52 via a drive shaft 53. Planet gears 54 mesh with the sun gear 52, and with a fixed annulus gear 55. Supported on the external surface of the fixed annulus gear 55 is an annular sleeve 56, from the interior surface of which projects a first movable annulus gear 57, which meshes with the planet gears 54. A second movable annulus gear 58 is received into an end part of the sleeve 56 remote from fixed annulus gear 55, the movable annulus gear 58 also meshing with the planet gears 54. In this arrangement, if the number of teeth of the fixed annulus ring 55 is n, and the number of planet gears 54 is two, and the numbers of teeth of the annulus gears 57 and 58 may be T, where T and n are integers,

$$T = n \pm 2k$$

and

$$0.9n < k < 1.1n$$

The direction of rotation of the annulus gears 57 and 58 is dependent on whether they have either more or less teeth than the annulus ring 55. The speed reduction ratio of the gearbox is dependent on the difference in the numbers of teeth between the fixed annulus gear and the movable annulus gears, and is also dependent on the

ratio of the numbers of teeth on the sun and planet gears. By choosing suitable numbers of teeth for the annulus gears 55, 57, and 58, the gearbox shown in figure 8 may provide a first drive output from gear teeth 59 of annulus gear 57, and a second drive output from stub shaft 60 attached to annulus gear 58, with both the direction and the speed of the two drive outputs being different from each other.

In the arrangement shown in figure 9, the planetary reduction gearbox of the present invention is used to provide a simple reversible drive. Figure 9 shows schematically a motor 60 driving a sun gear 62 via a shaft 63. Planet gears 64 mesh with the sun gear 62 and with a fixed annulus gear 65. A first movable annulus gear 66 is provided with fewer teeth than the annulus gear 65, and a second movable annulus gear 67 is provided with more teeth than the fixed annulus gear 65. If the differences in the numbers of teeth between annulus gears 65 and 66 and between annulus gears 65 and 67 are the same, and the output gear teeth 68 and 69 associated with annulus gears 66 and 67 respectively will rotate in opposite directions at equal speeds when the sun gear 62 is turned. An idler gear 70 mounted on a shaft 71 for axial sliding movement in bearings 72 and 73 transmits the output from the gearbox to a final drive gear 74. Sliding the shaft 71 and the idler gear 70 from the upper position shown in figure 9 to a lower position wherein

the idler gear meshes with the gear teeth 68 and the final drive gear 74 will cause the direction of the final drive gear 74 to be reversed while the motor 60 continues to rotate in the same sense. Figure 9 thus illustrates a compact and simple gear arrangement for reducing the speed of a motor and allowing reversing of the drive direction without the requirement to reverse the motor rotation direction.

Figures 10 and 11 shown schematically an alternative method of driving the planet gears of the gearbox. In figure 11, a motor 80 has a cranked drive shaft 81, on the offset portion of which a planet gear 82 is freely rotatable. The planet gear 82 meshes with a fixed annulus gear 83 and with a movable annulus gear 84, the two annulus gears having different numbers of teeth. The annulus gear 84 is formed with a depending flange 85 to provide bearing surfaces to support the annulus gear 84 on the outside surface of the annulus gear 83. Final drive from the gearbox may be taken by passing a drive belt 86 around the depending flange 85 as seen in figure 10, or alternatively by closing the end of the annulus gear 84 remote from the motor 80 and providing a stub shaft 87 with a coupling for connection to a final drive shaft.

Figures 12 and 13 are schematic perspective and sectional views, respectively, of a two-stage reduction gearbox

utilising the teachings of the invention. In the figures, a motor 90 drives a sun gear 91 via a drive shaft 92. The sun gear 91 meshes with two planet gears 93, which in turn engage the first fixed annulus gear 94 and a first movable annulus gear 95 having a different number of teeth from annulus gear 94. The external surface of the annulus gear 95 is formed with gear teeth 96, and this toothed surface constitutes a sun gear for a second planetary gear system. Outer planet gears 97 mesh with the gear teeth 96, and also mesh with an outer fixed annulus gear 98 and an outer movable annulus gear 99 having a number of teeth different from that of fixed annulus gear 98. The annulus gear 99 is formed with external gear teeth 100 for transmitting driving force to an external load. By a suitable selection of the diameters and numbers of teeth for the fixed and movable annulus gears, extremely large reduction ratios may be achieved without the need for excessively long trains of compound gears. Such large reductions may be useful in control systems for telescopes, where small movements are required, or to enable low-power motors to move large loads.

Referring now to Figures 14 to 16, there is seen a further planetary gearbox according to the invention. In figure 14, a sun gear 101, a planet gear 102, and two annulus gears 103 and 104 are shown in transverse section. The sun gear 101 has a central toothed portion

105, and smooth circular rolling surfaces 106 and 107 at its axial ends. The diameter d of the rolling surfaces 106 and 107 are arranged to be equal to the pitch circle diameter of the central toothed portion 105.

The planet gear 102 likewise has a central toothed portion 108, whose axial extent corresponds to the axial extent of the toothed portion 105 of the sun gear 101. At the axial ends of the planet gear 102, rolling surfaces 109 and 110 are formed with diameters corresponding to the pitch circle diameter of the toothed portion 108 of the planet gear.

The upper annulus gear 103 (as seen in the figure) is a formed with a toothed portion 111, and a smooth rolling surface 112 coaxial with the toothed portion and situated at an axial end thereof. The diameter of the rolling surface 112 is arranged to be substantially equal to the pitch circle diameter of the toothed portion 111. In the gearbox shown, annulus gear 103 has a toothed portion comprising 63 teeth.

The lower annulus gear 104 of figure 14 is formed with a toothed portion of 113 and a rolling surface 114. The toothed portion 113 of annulus gear 104 comprises 60 teeth, and the diameter of rolling surface 114 corresponds to the pitch circle diameter of the toothed portion 113 and is coaxial therewith. As in the previous

gearbox assemblies, the pitch circle diameters of the two annulus gears are substantially equal.

The gearbox is assembled by placing the sun gear and three planet gears symmetrically within the lower annulus gear 104, with the rolling surfaces 1100 of the planet gears 102 in contact with the rolling surface 114 of the annulus gear, and with the rolling surface 106 of the sun gear. The toothed portions 105 and 108 of the sun and planet gears are meshed together, and the lower (as seen in the figure) halves of the toothed portions 108 of the planet gears mesh with the toothed portion 113 of the annulus gear 104. It will be appreciated that, since the gear teeth of the toothed portions 105 and 108 of the sun and planet gears extend radially beyond the rolling surfaces 106 and 1100, relative axial movement between the sun and planet gears is prevented by contact between the axial end surfaces of the gear teeth and the axially-facing lands 106a, 110a extending between the root portions of the teeth.

To complete the assembly of the gearbox, annulus gear 103 is offered up coaxially with annulus gear 104, and oriented in rotation such that the teeth of the annulus gears 103 and 104 are in phase at their points of contact with the three symmetrically-positioned planet gears. This position is shown in figure 16, with the non-sectioned planet wheel omitted for clarity. Since the

difference in the numbers are teeth of the annulus gears is three, a symmetrical arrangement is reached wherein three planet gears are used to surround the sun gear. To perform a reduction drive, a driving rotation is applied to the sun gear, one of the annulus gears is held stationary, and the other annulus gear rotates at a reduced speed. External gear teeth 120 are shown in figure 15 to represent a means by which drive may be transmitted from the moving annulus gear. As before, the symmetrical arrangement of planet gears prevents any net radial force being applied to the sun gear. The rolling surfaces at the ends of the sun and planet gears, and on the annulus gears, serve to transmit any radial forces between the gears, thus relieving the gear teeth of any radial loading.

In an alternative arrangement (not illustrated), the rolling surfaces 106 and 107 of the sun gear may be positioned at the central region of the sun gear, with the toothed portion 105 separated into upper and lower toothed portions each adjacent an axial end of the sun gear. Similarly, the rolling surfaces 109 and 110 of the planet gears may be centrally positioned, and the upper and lower annulus gears 103 and 104 simply into changed so that the rolling surfaces 112 and 114 of the annulus gears are positioned centrally of the gearbox.

In a yet further alternative arrangement, rolling

surfaces may be provided only on the ring gears and planet gears, with alignment between the planet gears and sun gear being effected by engagement only of their respective gear teeth.

In all of the gearboxes of the present invention, it is preferable to enlarge the axial dimension of the sun and planet gears, and to utilise symmetrical arrangements of planet gears, in order prevent skewing of the axes of the planet gears and to balance lateral forces exerted on the sun gear. As a further means to eliminate asymmetrical forces, it is foreseen that the gearbox may be produced with two fixed annulus gears having equal numbers of teeth, and with the movable annulus gear positioned between the two fixed annulus gears. The planet gears would each engage both of the fixed annulus gears and the movable annulus gear. Engagement with the fixed annulus gears at the axial ends of each planet gear ensures that the axes of the planet gears remain in parallel alignment with the axis of the sun gear.

3. A planetary gearbox comprising a sun gear, a planet gear having a single series of teeth engageable with the sun gear, a fixed first internally-toothed ring gear having a first number of teeth and capable of meshing with the teeth of the planet gear, and a second internally-toothed ring gear of the same effective diameter as the first ring gear and capable of meshing with the teeth of the planet gear, but having a number of teeth different from that of the first ring gear, and wherein:

the planet gear includes rolling surfaces coaxial with and having substantially the same diameter as the pitch circle of its teeth; and

each of the ring gears includes a rolling surface coaxial with and having substantially the same diameter as the pitch circle of its teeth;

and wherein each of the rolling surfaces of the ring gears engages with a rolling surface of the planet gear, and the arrangement being such that when a drive input is applied to the sun gear, the planet gear travels around the sun gear in meshing engagement with both of the ring gears and the second ring gear rotates relative to the first ring gear.

4. A planetary gearbox according to claim 3, wherein the number of teeth of the first ring gear is less than the number of teeth of the second ring gear.

5. A planetary gearbox according to claim 3, wherein the number of teeth of the first ring gear is more than the number of teeth of the second ring gear.

6. A planetary gearbox according to any of claims 3 to 5, wherein a plurality of planet gears surround the sun gear at equal angular spacings.

7. A planetary gearbox according to claim 6, wherein the difference between the number of teeth of the first ring gear and the number of teeth of the second ring gear is the same as the number of planet gears in the gearbox.

8. A planetary gearbox according to claim 6, wherein the difference between the number of teeth of the first ring gear and the number of teeth of the second ring gear is a multiple of the number of planet gears in the gearbox.

9. A planetary gearbox according to any of claims 3 to 8, wherein the difference between the number of teeth of the first ring gear and the number of teeth of the second ring gear is equal to or less than one tenth of the number of teeth of the first ring gear or the number of teeth of the second ring gear.

10. A planetary gearbox according to any of claims 3 to 9, further comprising a third ring gear coaxial with the

first and second ring gears of the same effective diameter as the first and second ring gears and capable of meshing with the teeth of the planet gear.

11. A planetary gearbox according to claim 10, wherein the third ring gear has the same number of teeth as the first fixed ring gear and is held stationary, and wherein the second ring gear is positioned between the first and third ring gears.

12. A planetary gearbox according to claim 10, wherein the third ring gear has a different number of teeth from the first fixed ring gear, and is rotatable relative thereto.

13. A planetary gearbox according to claim 12, wherein the difference between the numbers of teeth of the first and second ring gears is the same as the difference between the numbers of teeth of the first and third ring gears.

14. A planetary gearbox according to claim 13, wherein the second ring gear has more teeth than the third ring gear.

15. A planetary gearbox according to claim 13, wherein the second ring gear has less teeth than the third ring gear.

16. A planetary gearbox according to any of claims 12 to 15, wherein the second and third ring gears are provided with means to transmit driving force to an external transmission element.

17. A planetary gearbox according to claim 16 wherein the means to transmit driving force comprises a pulley, a sprocket, or a set of external gear teeth formed on the ring gear.

18. A planetary gearbox according to any preceding claim 16 wherein the combined axial lengths of the ring gears substantially equal the respective axial lengths of the sun and the planet gears.

19. A two-stage epicyclic reduction gearbox, comprising a sun gear, number of inner planet gears, first and second respectively fixed and movable ring gears of equal diameter but differing numbers of teeth, the second ring gear being externally toothed, a number of outer planet gears the which mesh with external teeth of the second ring gear and with third and fourth respectively fixed and movable ring gears, the third and fourth ring gears being arranged co-axially with the first and second ring gears, the third and fourth ring gears being of equal diameter and having differing numbers of teeth and the fourth ring gear having external drive means.

20. A two-stage epicyclic reduction gearbox according to claim 19, wherein the number of inner planet gears is the same as the number of outer planet gears.

21. A two-stage epicyclic reduction gearbox according to claim 19, wherein the number of inner planet gears is more than the number of outer planet gears.

22. A two-stage epicyclic reduction gearbox according to claim 19, wherein the number of inner planet gears is less than the number of outer planet gears.

23. A planetary reduction gearbox substantially as described herein with reference to figures 1 to 3, figure 4, figure 5, figure 6, figure 7, figure 8, figure 9, figures 10 and 11 or figures 12 and 13 of the accompanying drawings.

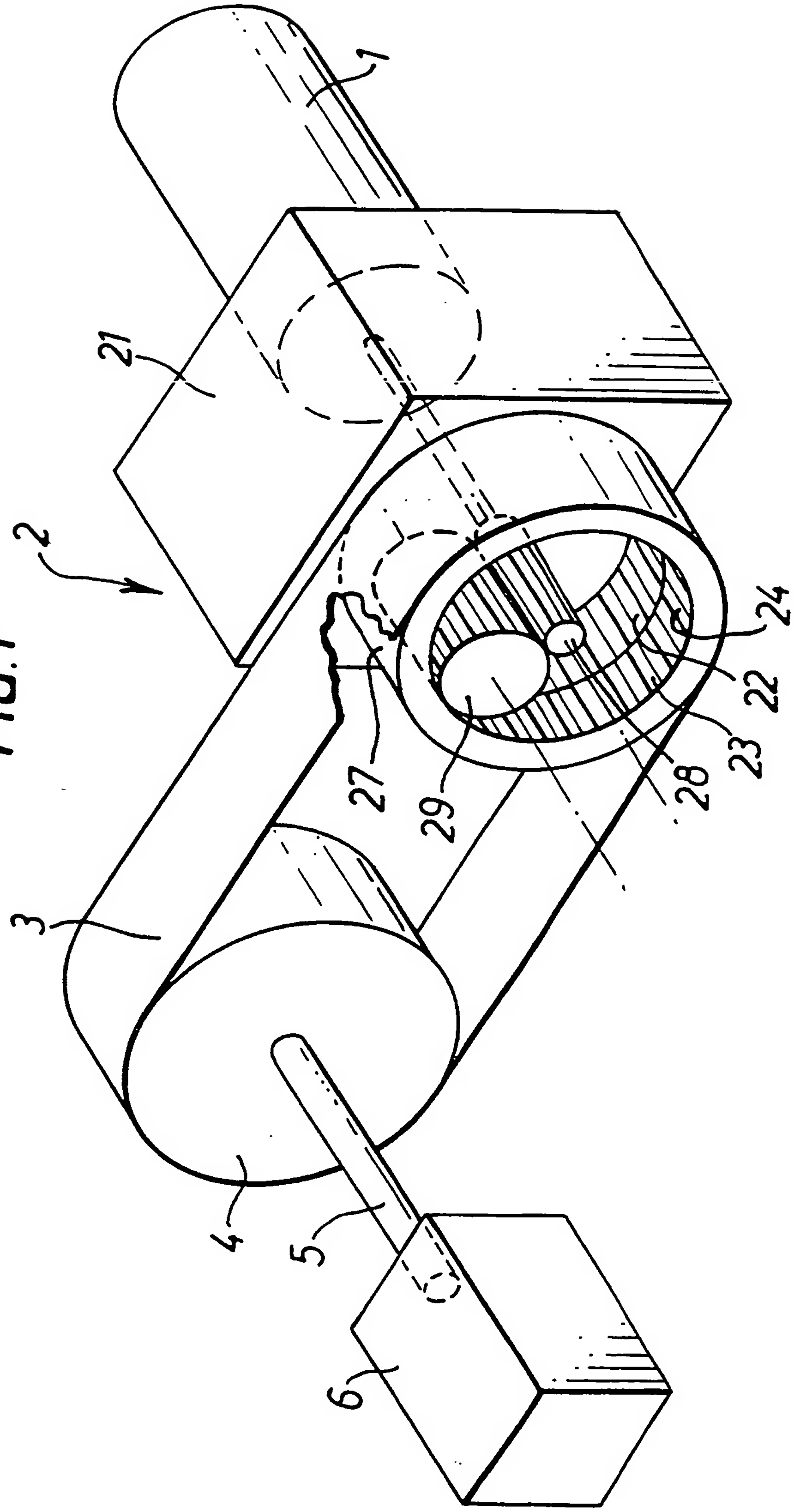


FIG.1

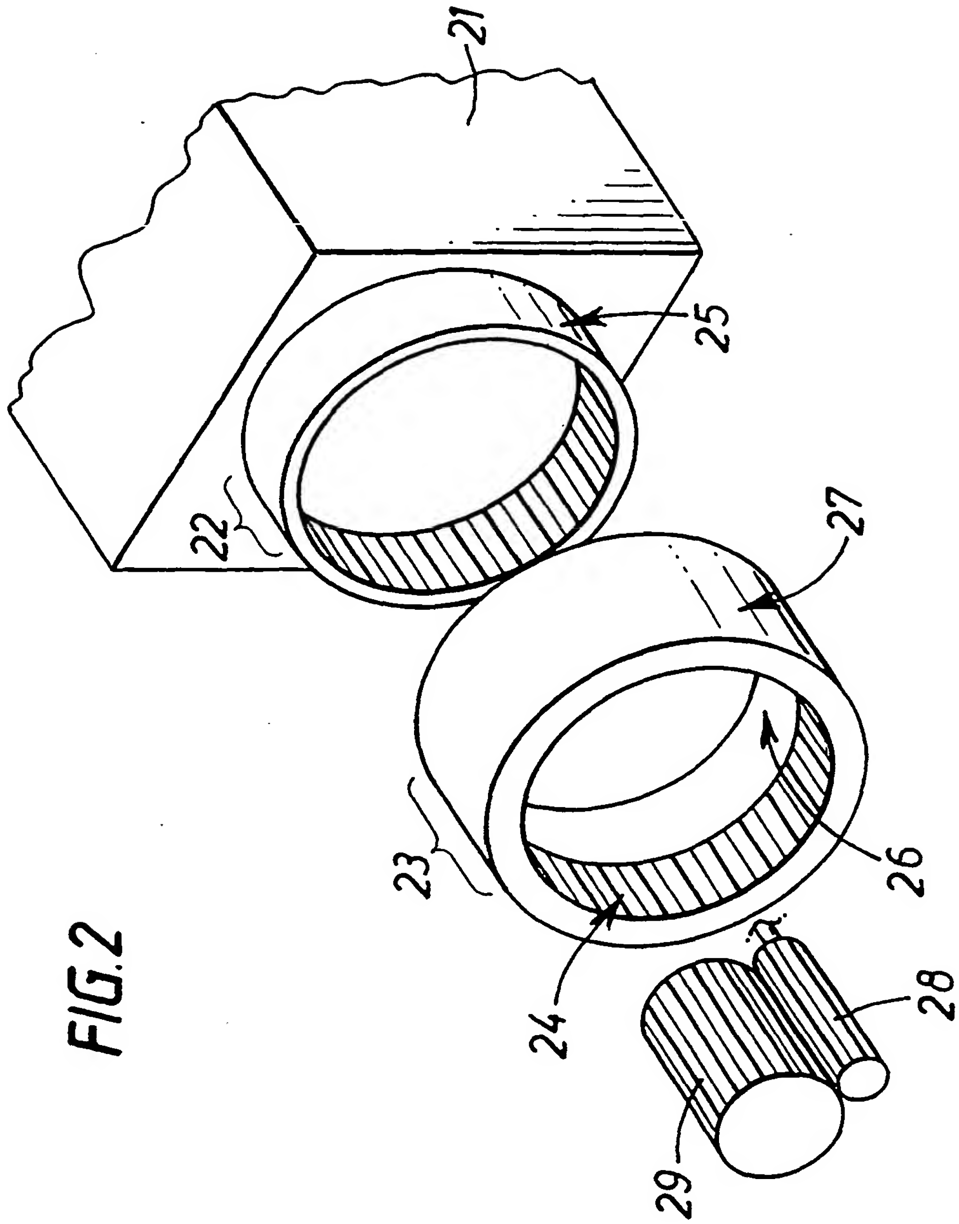


FIG.2

FIG. 3

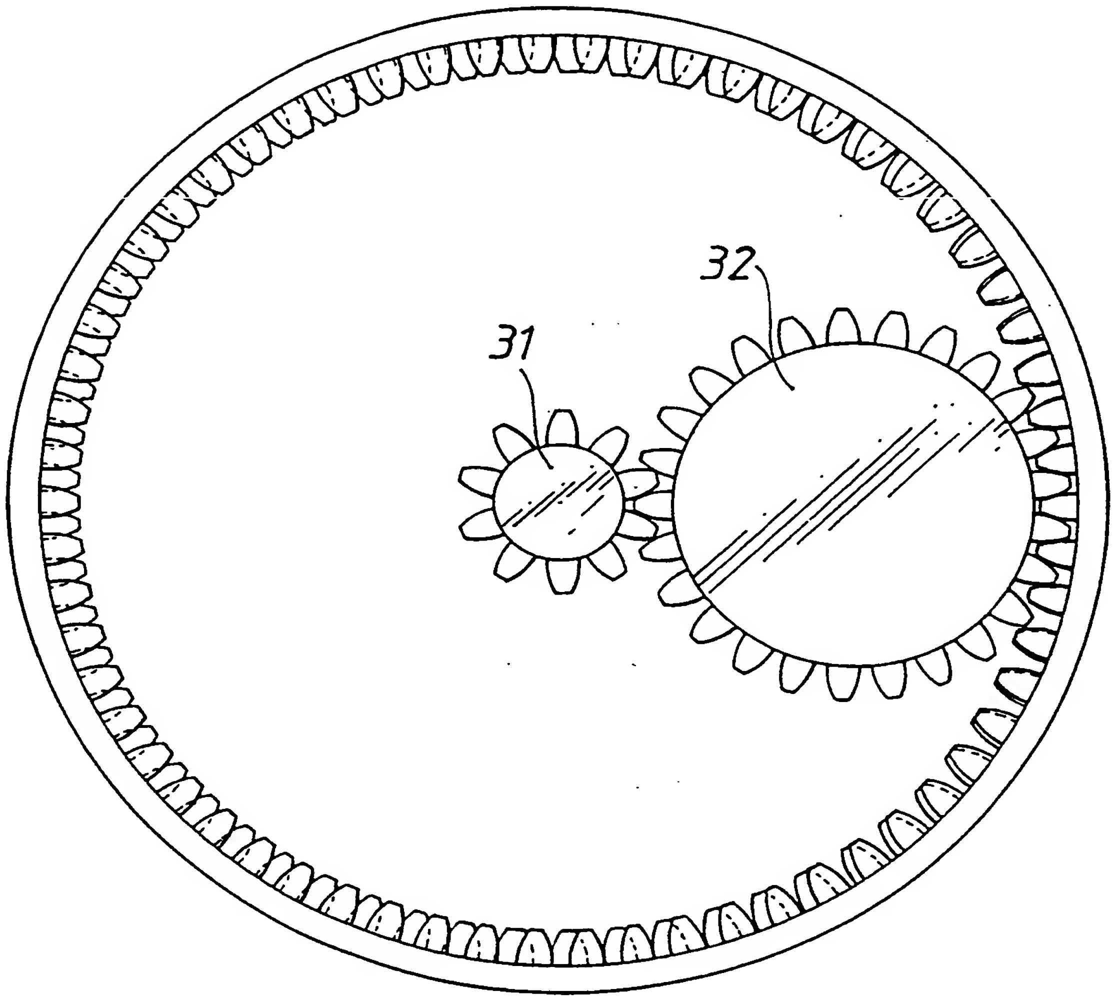


FIG.4

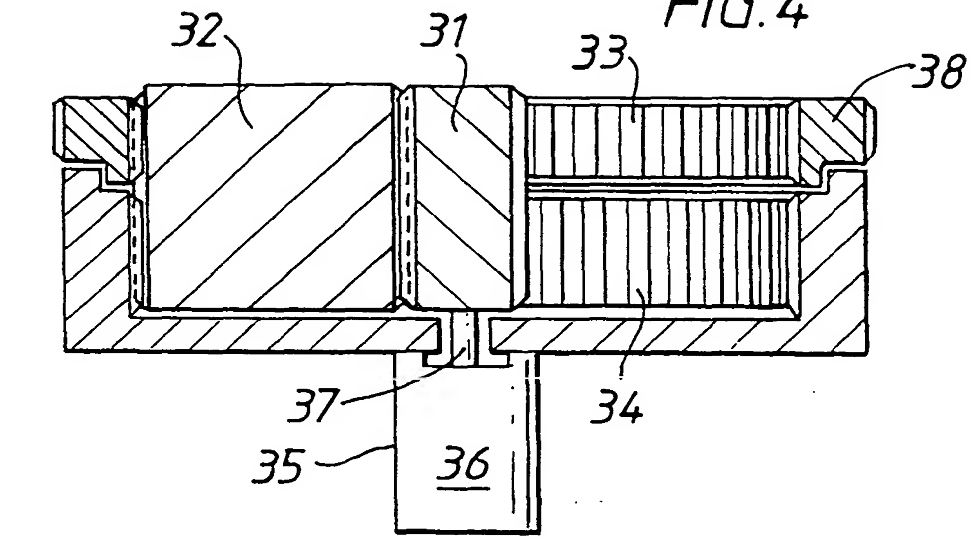
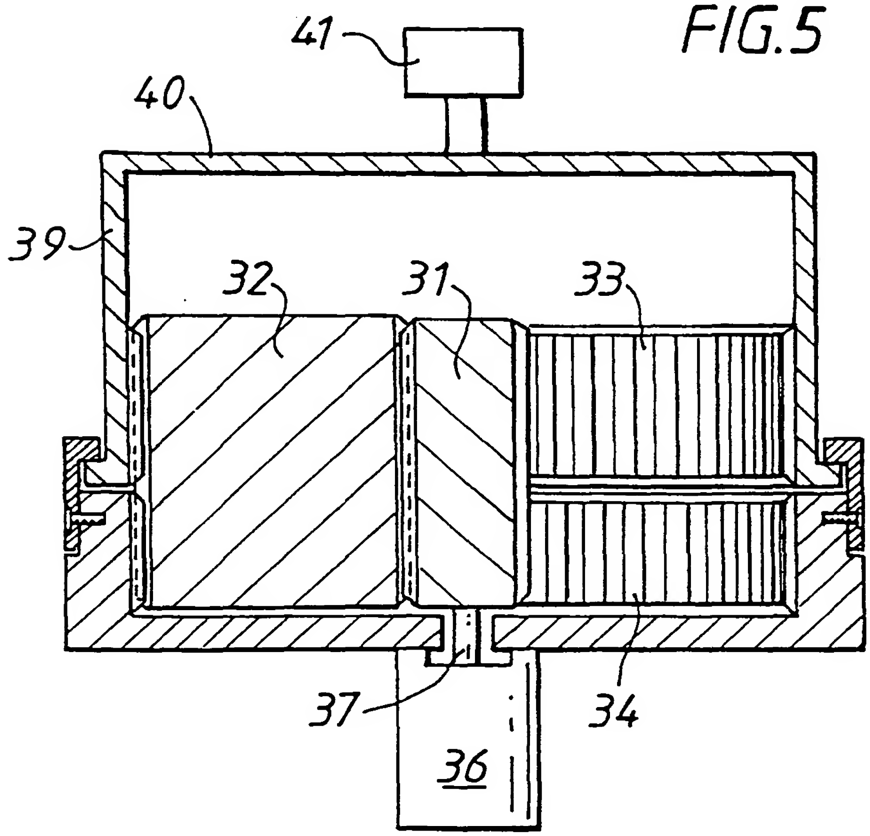


FIG.5



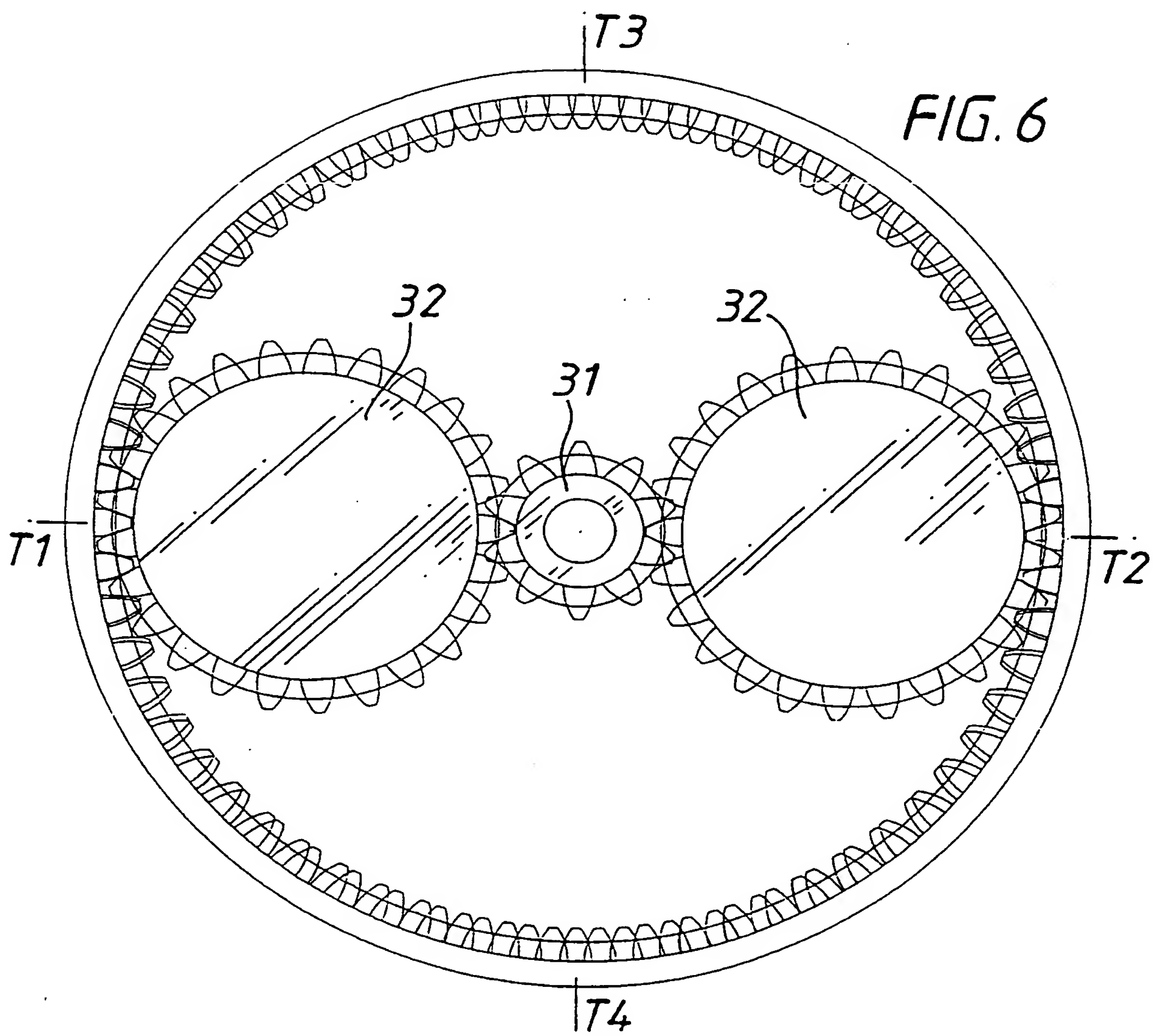


FIG. 6

FIG.7

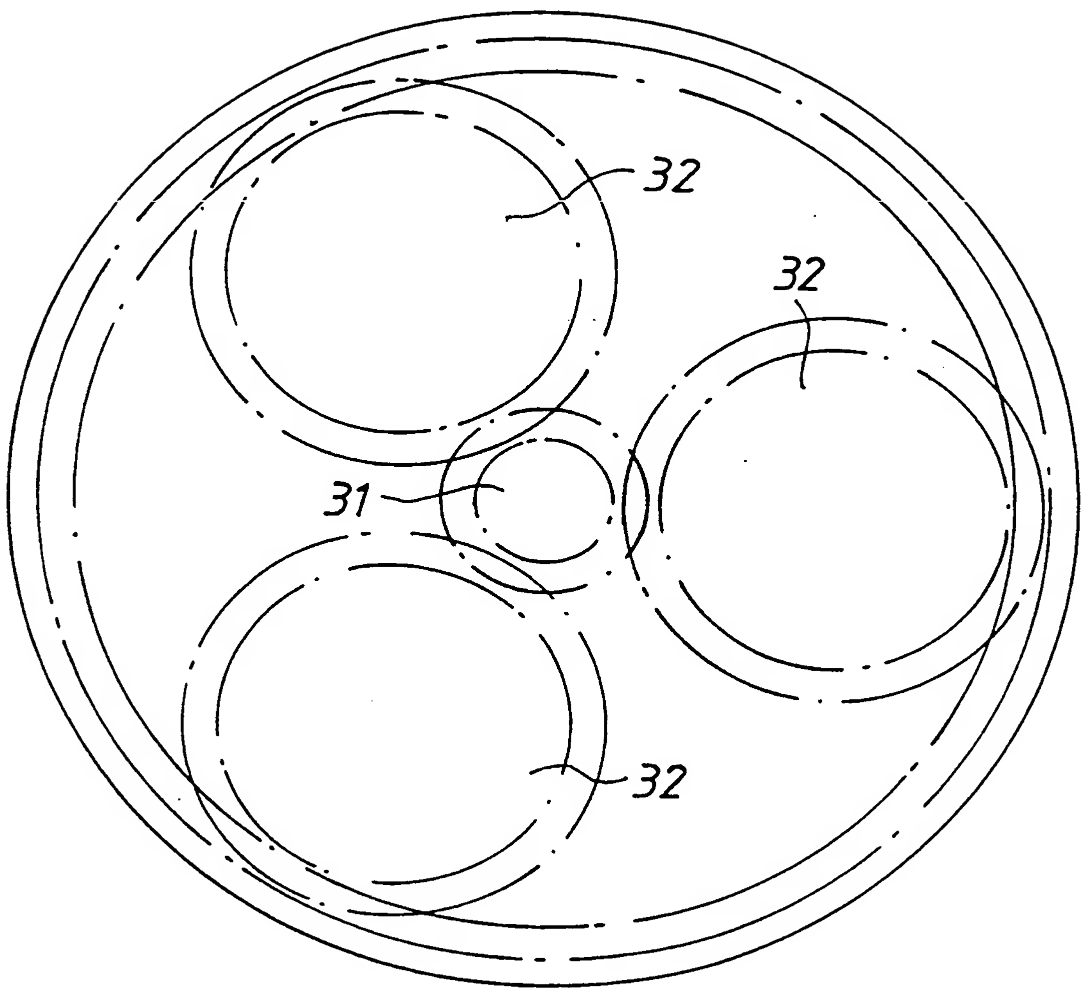


FIG.8

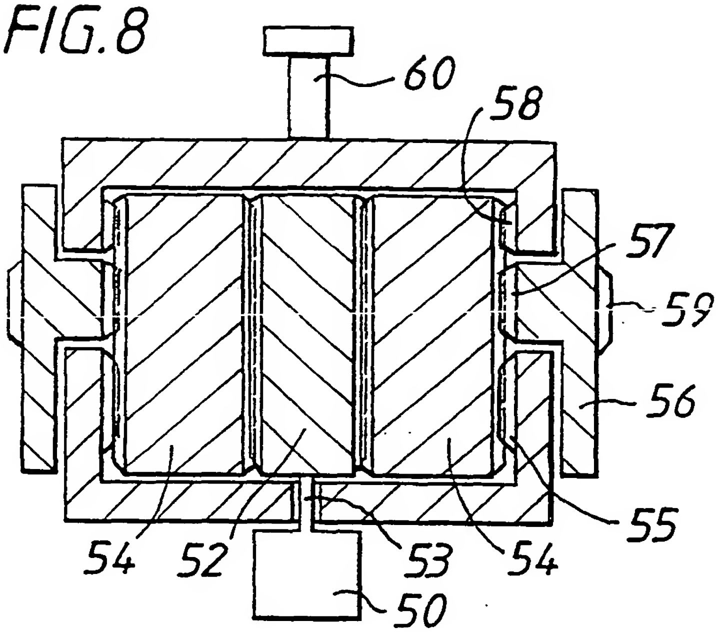

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FIG.9

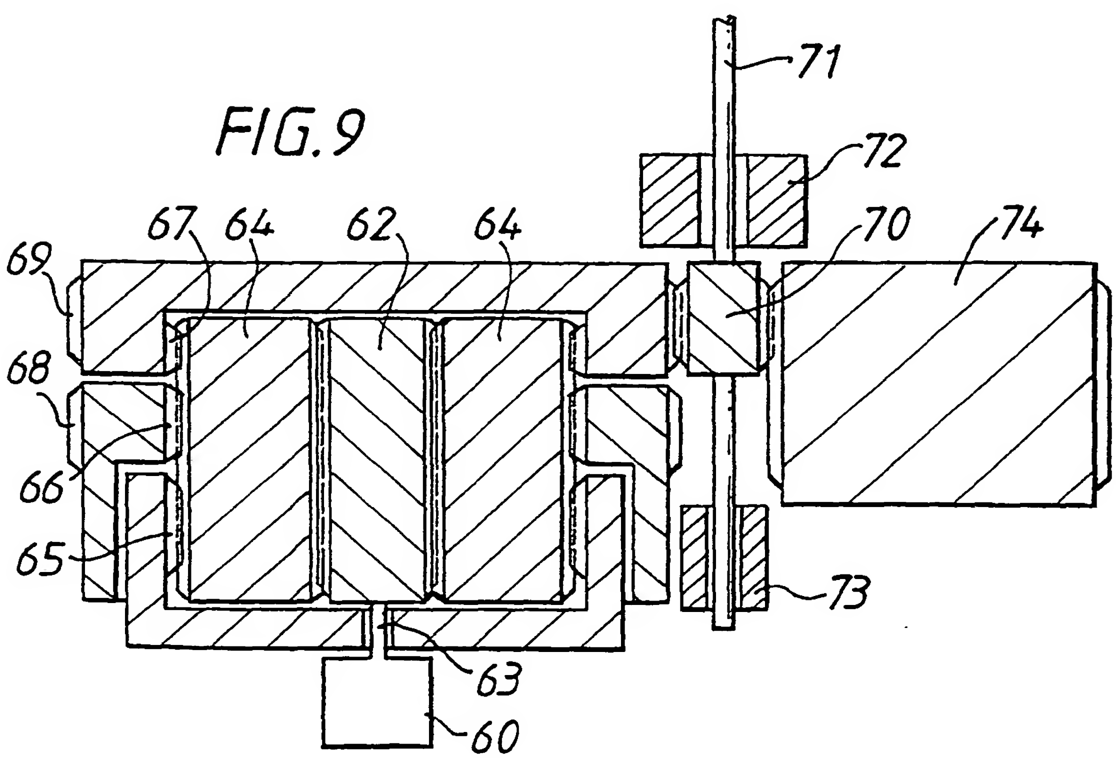
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FIG. 10

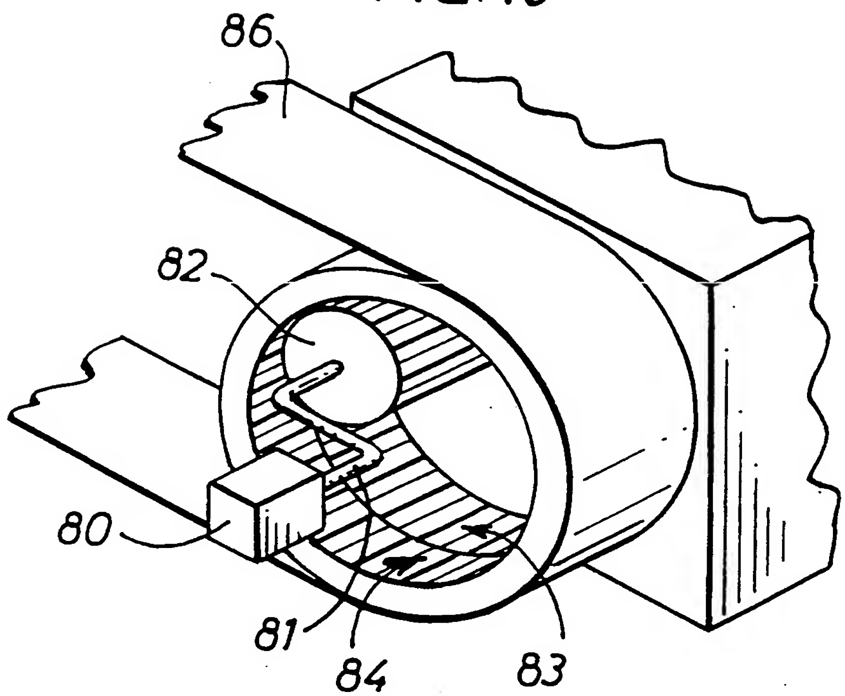


FIG. 11

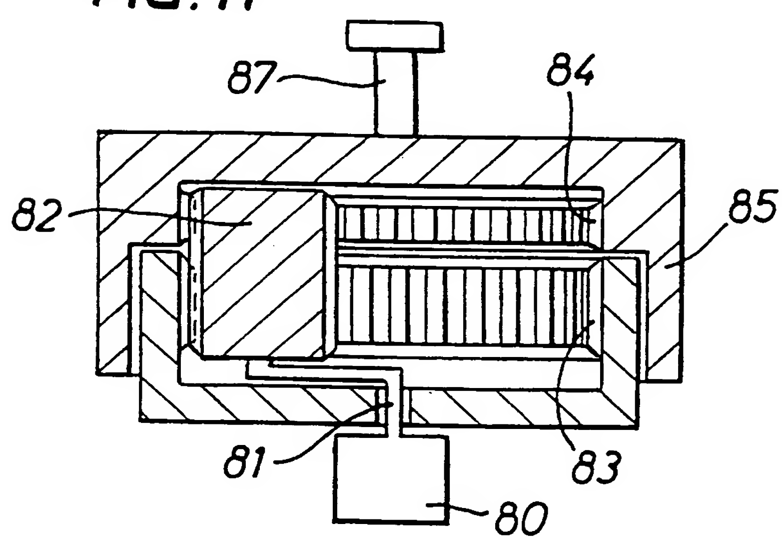


FIG. 12

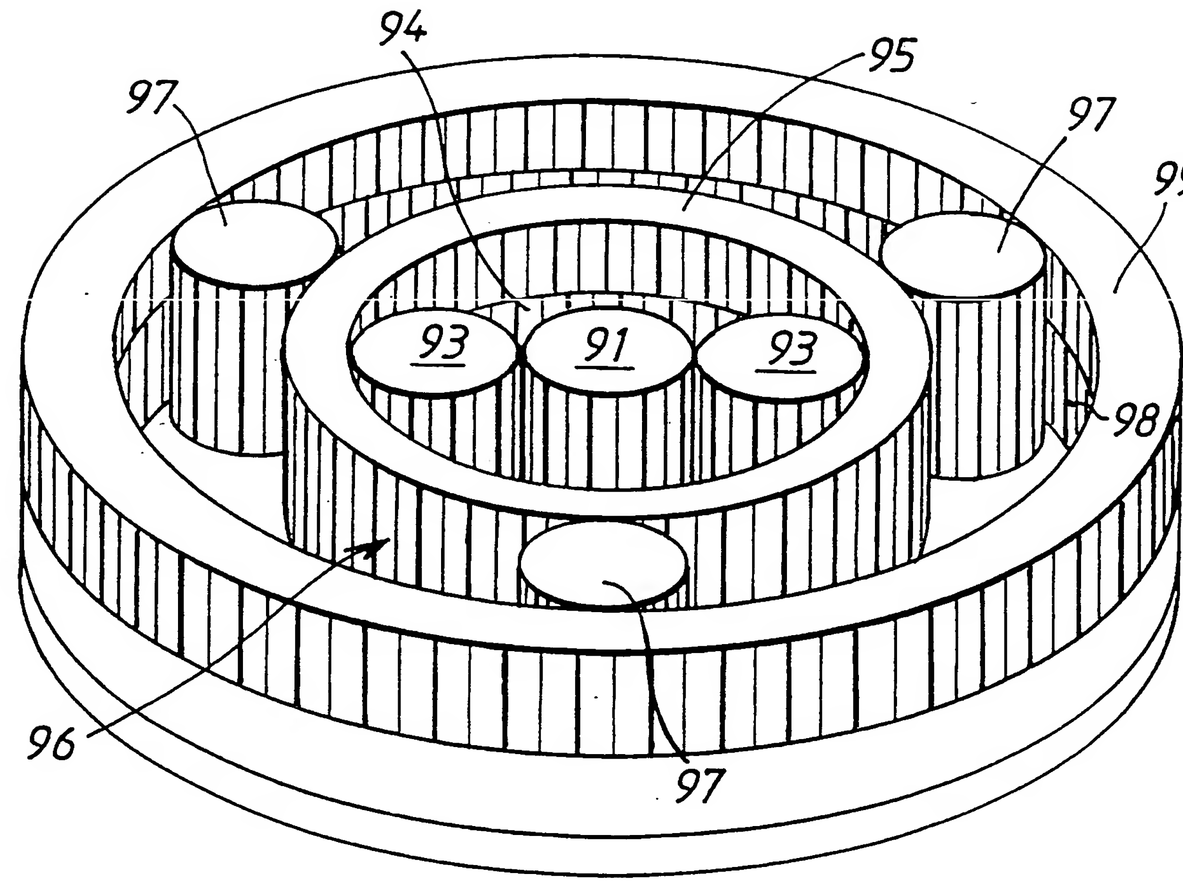
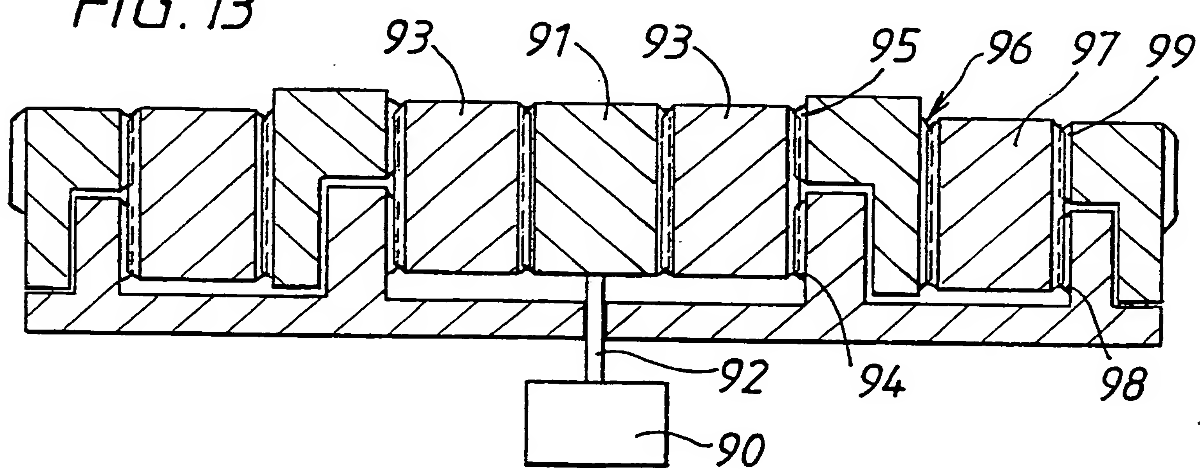


FIG. 13



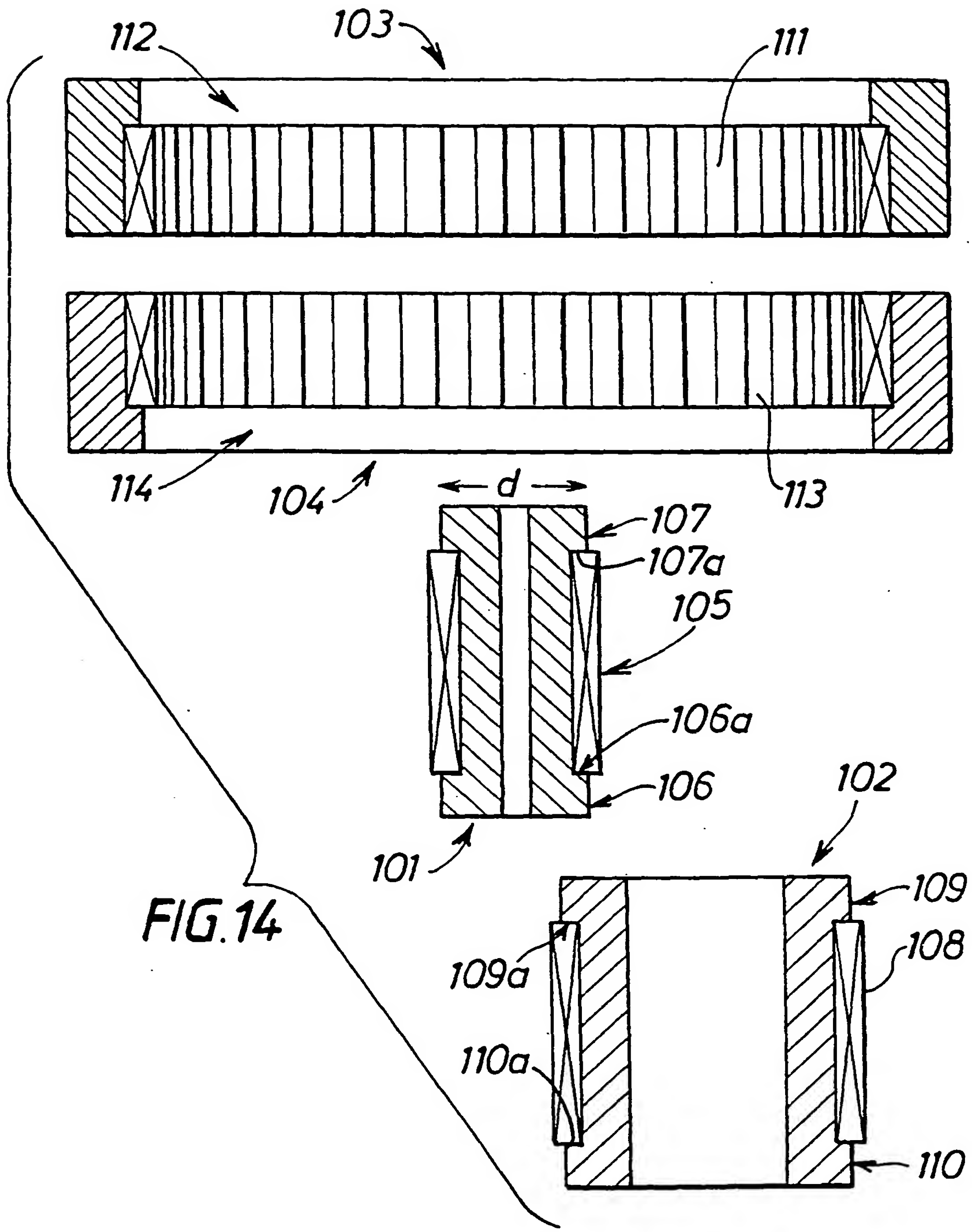

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FIG.14

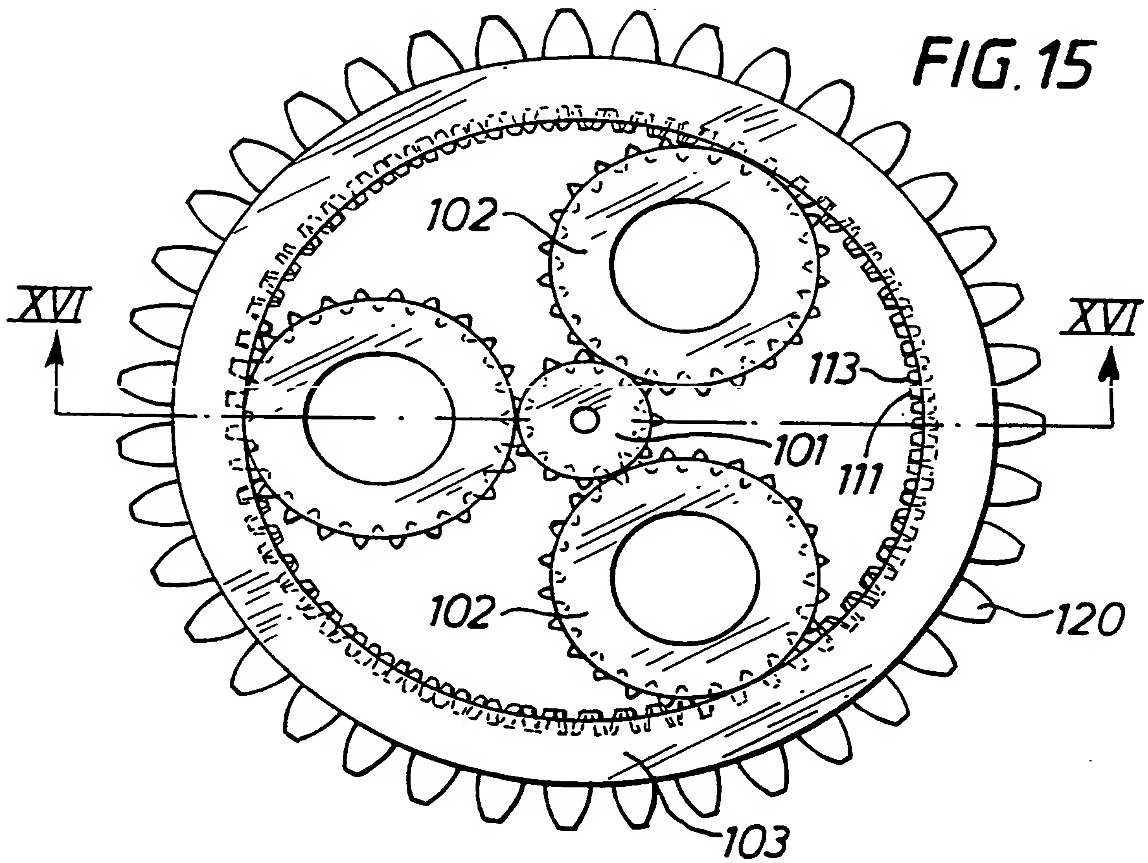

FIG. 15

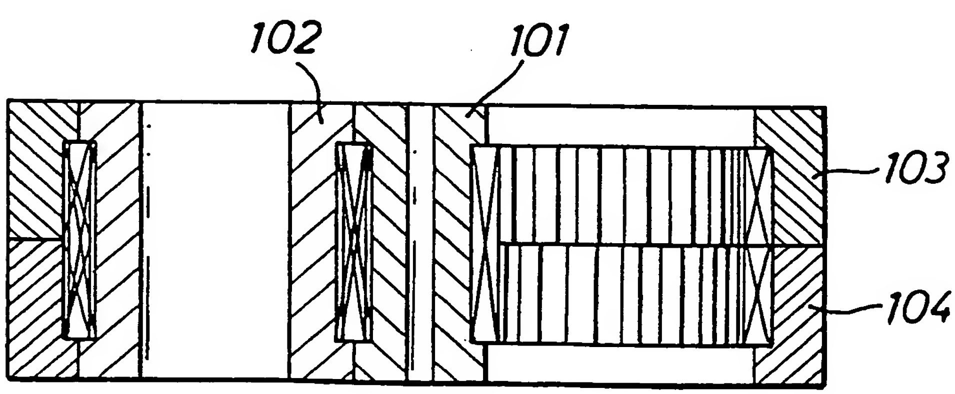

FIG. 16

INTERNATIONAL SEARCH REPORT

International Application No
PCT/GB 00/01474

A. CLASSIFICATION OF SUBJECT MATTER
IPC 7 F16H1/46 F16H57/02

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 7 F16H

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

EPO-Internal, WPI Data

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	US 4 850 247 A (YU) 25 July 1989 (1989-07-25) column 7; figures 1-5	1-7
A	US 1 425 430 A (WIKANDER) 8 August 1922 (1922-08-08) page 1; figure 1	1-3,6

☐ Further documents are listed in the continuation of box C.

☒ Patent family members are listed in annex.

° Special categories of cited documents :

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"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
25 July 2000	01/08/2000

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL – 2280 HV Rijswijk Tel. (+31-70) 340-2040, Tx. 31 651 epo nl, Fax: (+31-70) 340-3016	Flores, E

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INTERNATIONAL SEARCH REPORT

Information on patent family members

International Application No
PCT/GB 00/01474

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
US 4850247 A	25-07-1989	NONE	
US 1425430 A	08-08-1922	NONE	